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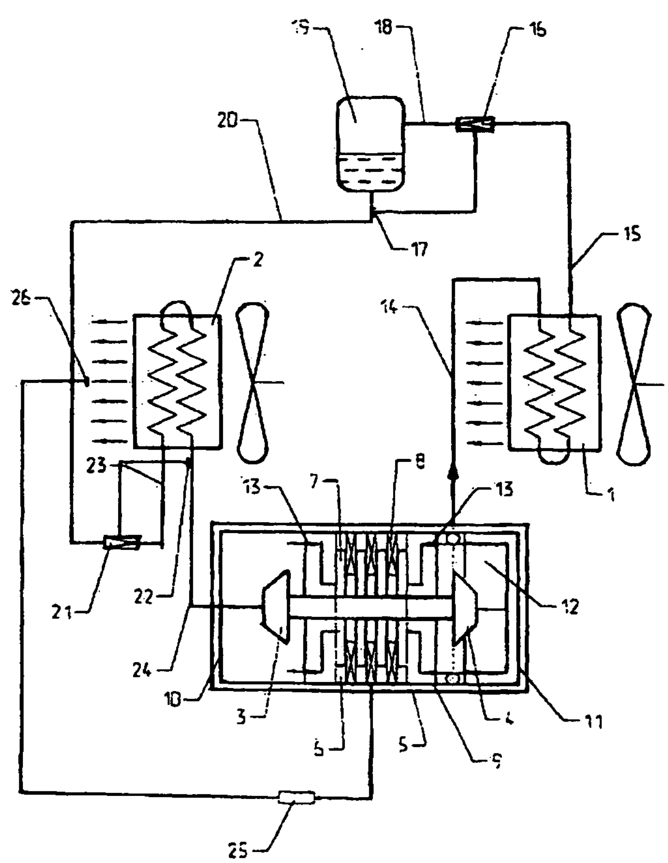
(54) Title: REFRIGERATING SYSTEM WITH AN INTEGRATED TURBOCOMPRESSOR



(57) Abstract: Refrigerating system comprising a condenser (1), an expansion valve (21), an evaporator (2) and a two-stage centrifugal compressor assembly (5) with a built-in valve disk collectorless electric motor (6) whereby the refrigerant gas flowing from the first stage (3) of the compressor to the second stage (4) of the compressor passes through spaces in the motor (6) thereby cooling the motor (6), the expansion valve (21) being regulated by the pressure and the temperature of the refrigerant leaving the evaporator (2) and the speed of the motor (6) being regulated by the temperature of the air oat the exit of the evaporator (2)..

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REFRIGERATING SYSTEM WITH AN INTEGRATED TURBOCOMPRESSOR

FIELD OF THE INVENTION

The field of the invention is refrigeration technology, more particularly, turbocompressors.

BACKGROUND OF THE INVENTION

There is a known refrigerating system comprising a condenser, an evaporator, and a two-stage centrifugal compressor assembly with a built-in electrical motor between the stages. The motor is provided with a cooling system that has coolant feed and discharge ducts connected to the internal cavity of its housing. Further, the condenser inlet is connected to the compressor assembly outlet, and the condenser outlet is connected by one line to the evaporator inlet via the first choke element, and by a second line to the electrical motor cooling system via the second choke element. The evaporator outlet communicates with the compressor assembly inlet, and the gas coolant discharge duct is connected to the outlet of the compressor assembly's second stage (1).

The drawbacks of this known cooling system are low efficiency and reliability due to the fact that wet vapor can enter the exhaust of the compressor's first stage; the presence of oil in the system; the narrow range of controllability; and the fact that ecologically safe coolants cannot be used.

There is a known centrifugal compressor assembly for a refrigerating system comprising two centrifugal compressor stages with rotors, diffuser ducts, collection chambers, inlet and outlet tubing for the second stage, and a built-in electrical motor between the compressor assembly's first and second stages. The rotor of the electrical

motor is positioned with the centrifugal stage rotors on one shaft, which is mounted on bearings, and the stator is fixed inside its housing, forming a cooling skirt between them. Further, the motor housing is provided with gas coolant feed and discharge ducts which are connected to the outlet tubing of the compressor assembly's second stage (1).

The drawbacks of this known centrifugal compressor assembly for a cooling system are low reliability due to the unsatisfactory design of the electrical motor's cooling system; the lack of total compensation for axial forces on the assembly's shaft; the large size of the electrical motor; and the fact that basic design solutions cannot be used for small assemblies, such as those with 5 and 8 kW cooling capacities.

US 6,070,421 (Verechtchagin et. al,) relates to a refrigerating system that has a condenser, an evaporator, and a two-stage centrifugal compressor assembly with an electric motor provided with a cooling system. A condenser inlet is connected to a compressor assembly outlet, while a condenser outlet is connected via first and second choke elements to an inlet of the evaporator and to the cooling system. The evaporator outlet communicates with the compressor assembly inlet, and the cooling system outlet communicates with the inlet of the second stage of the assembly. The cooling system is provide with a cooling skirt having a cavity of the cooling system lying between a stator and a housing. The refrigerating system has a separator vessel, a recuperative heat exchanger, and a heat-regulating valve controlled by the coolant pressure and temperature at the cooling skirt inlet, and also fitted in series with the separator vessel in the line connecting the condenser inlet to the evaporator. The separator vessel is connected by a gas phase to the coolant feed duct. The heat exchanger is connected by the coolant to a line connecting the condenser outlet to the evaporator inlet via the separator vessel. The second choke element is a heat-controlled valve regulated by the pressure and temperature of the coolant in the coolant discharge line from the cooling skirt.

This system shows a number of disadvantages related to the efficiency and the cooling system.

Thus a need for an improved turbocompressor remains.

SUMMARY OF THE INVENTION

According to one aspect, the present invention relates to a refrigerating system comprising: an evaporator, a condenser, two diffusers, and a two stage centrifugal

compressor assembly with a built-in valve disk collectorless motor between the compressor stages.

According to another aspect, the present invention relates to a refrigerating system comprising: an evaporator, a condenser, two diffusers, and a two stage centrifugal compressor assembly with a built-in valve disk collectorless motor between the compressor stages; wherein the flowing gas flowing from the first stage of the compressor to the second stage of the compressor, flows through spaces in said motor, and thus effectively cools said motor.

According to another aspect, the present invention relates to a refrigerating system as defined above, wherein said system is equipped with a drosseling valve, which controls the temperature and the pressure of the cooling agent upon leaving the evaporator.

According to another aspect, the present invention relates to a refrigerating system as defined above, which is automated in such a way that provides a smooth start and stop of the compressor.

According to another aspect, the present invention relates to a refrigerating system as defined above, which is equipped with means of changing the engine speed which controls the temperature of the air on the exit of the evaporator.

According to another aspect, the present invention relates to a refrigerating system comprising: an evaporator, a condenser; two diffusers; and a two stage centrifugal compressor assembly with a built-in valve disk collectorless motor between the compressor stages; said sprockets and diffusers designed to improve their efficiency.

According to another aspect, the present invention relates to a novel, efficient sprocket and diffuser.

BRIEF DESCRIPTION OF THE DRAWINGS

Fig. 1 represents a scheme of an air conditioning system.

Fig. 2 represents a longitudinal sections of the turbocompressor of the present invention

Fig. 3 represents thumbnails of a blading section (sprocket and diffuser).

Figs. 4, 5, 6, and 7 represent an illustration of the motor, mainly the rotor. Note, the numerals in Figs. 4-7 do not necessarily comply with the numerals in figs. 1-3. In the event of a contradiction between the two in the specification, the latter should be considered.

DETAILED DESCRIPTION OF PREFERRED EMBODIMENTS

The system of air conditioning comprises: the condenser, evaporator and centrifugal compressor with the built-in electric motor equipped with an effective system of cooling of the engine. The cooling system is executed inside the compressor casings. The entrance of the condenser is bridged to the exit of the compressor, and the exit of the condenser through the first and second drosselling valves is attached to the entrance of the evaporator. The exit of the evaporator is connected with the compressor's inlet. Thus, the closed refrigerating system is created. The air conditioning system includes a vessel-separator. The vessel-separator is placed on a main line connecting the exit of the condenser to the entrance of the evaporator, between the first and second drosselling valves.

The turbocompressor of the present invention can be utilized in industrial and household air conditioning systems, and also in low-power chillers.

The described air conditioning system consists of a condenser, an evaporator and a two-stage centrifugal compressor with a built-in disk electric motor installed directly on the shaft of the compressor between the stages. The engine is equipped with a cooling system. The cooling of the electric motor utilizes the total flow of the cooling agent, pumped through shank bore of the compressor through special channels made in the body and bearing shields. Thus, the gas, leaving the first stage of compression, moves directly into the shank bore of the compressor, and therefrom passes to the entrance of the second stage of compression. Further, leaving the second stage of compression, the gas enters the entrance of the condenser. After the condenser, the cooling agent through the first, second drosselling units and the vessel-separator moves on to the entrance of the evaporator. The exit of the evaporator is bridged to the compressor's inlet. Thus, the scheme of vapor-compression cycle of two-stage compression with twofold throttling and incomplete intermediate cooling is implemented in the system.

The prior attempts of implementation of such scheme of cycle for the centrifugal compressor in the field of small and middle range of cooling had low efficiency in comparison to the widespread cycles utilized in the compressors of volumetric type (piston, propeller etc.). It was due both to low efficiency of the compressor, and the usage of inefficient electric motors. Besides the above mentioned, the padding losses were introduced to the cycle by an inefficient cooling system, oil system and friction losses in external pipe lines.

The described centrifugal compressor included in the structure of an air conditioning system, consists of two stages of compression installed in opposition on one shaft; two diffusers; two covers - collectors of gas; details forming internal passages for cooling agent transfer; two radial graphite seals; bearing assemblies; input and output pipe lines; the built-in disk electric motor set on the shaft of the compressor; a stator fixed in the compressor casing.

Prior attempts to utilize centrifugal compressors in the range of small and middle cooling output (5 - 100 kWt) turned out to be unsuccessful. The compressors had low efficiency because of the absence of ways of contouring of blading sections (sprockets and diffusers) of small productivity. The efficiency of a blading section in producing the required pressure ratio at the small consumption did not exceed 40 %. Hence there was large heat release, which lowered the efficiency of all the cycle.

The rather low efficiency of the utilized electric motors (for example, asynchronous type) resulted in the necessity to create unwieldy cooling systems. This resulted in the complexity of the cycle scheme, increase of weight and overall dimensions.

The absence, until recently, of the precision high-speed rolling-contact bearings (up to 50000 rev/min) made necessary the utilization of gas-static rolling-contact bearings. Such bearings require the presence of a composite system of feeders and an additional pressure source.

The usage of turbocompressor of the present invention allows to troubleshoot the problem of transition to environment-friendly ozone-safe cooling agents, simultaneously, guaranteeing, as a minimum, the same efficiency (COP) that exists with the present systems that utilize the forbidden and transient cooling agents (Montreal Protocol). Simultaneously, the overall dimensions and weight of the system is decreased (up to 20 %) at the expense of reducing the overall dimensions of heat exchangers and exclusion of the oil system. The decrease of the overall dimensions of heat exchangers is possible because of absence of oil in the system: there is no necessity to expand the area of heat interchange. The oil does not hinder the process of heat interchange. In addition to the above, the absence of low frequency noise simplifies the problem of sound insulation.

Due to frequency start and variable-frequency control of the compressor, the system has no starting currents and, most important, the control band of cooling output comprises 20 - 100 %.

The air conditioning system with the pressurized integral centrifugal compressor stated in the given patent, consists of the condenser, evaporator and a two-stage centrifugal compressor with the built-in disk electric motor installed between the stages. The compressor has a cooling system of the electric motor by a total flow of gas executed inside the compressor. The exit of the compressor is bridged to an entrance of the condenser. The exit of the condenser through two drosselling valves is attached to the entrance of the evaporator. The exit of the evaporator is bridged to the compressor inlet. We offer to utilize the scheme of cycle with two-stage throttling, incomplete intermediate cooling and to create a cooling system of the electric motor (built-in in a cavity of the compressor) by means of the total flow of gas. The supply system of a cooling agent in shank bore of the compressor is formed by channels made directly in the body, shields of bearings and is specially constructed of the insert of the second stage. Through these channels, the gas leaves the first stage of compression and moves directly into the cooled cavity, and then to the entrance of the second stage of compression.

The vessel-separator, in which the incomplete intermediate cooling implements, is set on the main line connecting the exit of the condenser to the entrance of the evaporator, between the two drosselling valves. The drosselling valves are, in the essence, thermostatic expansion valves (TEV), which control the pressure and temperature of the cooling agent passing through them, opening and being closed at a returning signal that comes from sensors. The sensor of the first drosselling valve is established directly on the exit of a drosselling unit. It controls intermediate pressure in cycle P *, which is determined by value calculated by conventionally for cycles with two-stage compression and incomplete intermediate cooling. The sensor of the second drosselling valve is set on the main line of the exit of the cooling agent from the evaporator. It dispatches a signal for opening or closing of the drosselling unit depending on value of overheating of the cooling agent in the evaporator. The overheating of the cooling agent is a rated value defined through the thermodynamic calculation of cycle. The vessel-separator is necessary for separation of the vapor-liquid mixture, which enters it after the first throttle into a coolant fluid and gas phase.

The air conditioning system is equipped with the device for the control of the speed of rotation of the engine depending on the temperature of the air coming out of the evaporator. Magnifying or reducing revolutions, we, thus, change the compression ratio of the compressor, implementing a necessary difference of pressure dependent on a difference

between temperature of evaporation and condensing temperature. Thus, depending on the temperature of ambient air we can set necessary compressor speeds (compression ratio). It allows to select an optimum regime for real operation conditions. To sum it all up, it is possible to tell, that the presence of two control modes: according to the consumption of the cooling agent (TEV is implemented) and according to a compression ratio (by changing the rotation frequency) allows to expand essentially and to optimize the range of operation of the air conditioning system with the centrifugal compressor. This is an essential advantage compared to the air conditioning systems with compressors of a volumetric type, that utilize a constant value of the compression ratio and vary only the consumption of a cooling agent. The range of change of the consumption is limited to the geometrical sizes of a drosselling system of the air-conditioner (capillary tube or TEV).

The centrifugal compressor included in an air conditioning system, consists of: two stages of compression installed in opposition on the unified shaft; two diffusers; two covers - gas collectors; details forming internal passages for the transfer of the cooling agent; two radial graphite seals; bearing assemblies; input and output of pipe lines; the built-in disk electric motor set on the shaft of the compressor; a stator fixed in the compressor casing. The first and second stage of the compressor are bridged among themselves directly through channels in shank bore of the compressor, thus the external connection pipes are eliminated. The disk electric motor is installed between the stages. The disks of the rotor are set on the shaft of the compressor between the stages. The shaft rests on bearing assemblies assembled on the basis of radial thrust bearings of rolling. The stator is captured in the compressor casing and is set in relation to the shield of the bearing through a special assembly detail. In the rotor and stator disks, and also in the stator frame the special channels for passing of gas are made. The axle-load on the shaft of the compressor produced by the gas-dynamic of forces, is partially compensated for by a special design of seals. The shaft is set free from thrusts with the help of advanced load executed at assembly on a floating support. The preliminary load is formed by spiral pressured springs.

Due to the utilized scheme of cooling, the thermal currents are removed not only from the electric motor, but from clusters of friction (radial thrust bearings) as well. The utilization of graphite seals in radial inserts allowed to essentially lower the heat production in them.

Air conditioning system (Fig.1) consists of the condenser 1, evaporator 2 and a two-stage centrifugal compressor (5) with the built-in disk electric motor (6) between stages (3) and (4). The compressor (5) has a built-in cooling system of the electric motor (6). The cooling of the engine (6) utilizes the total rate of gas flow that is sucked through the shank bore of the body (9) of the compressor (5). The exit of the compressor, main line (14), is bridged to the entrance of the condenser (1). From the condenser (1) the overcooled vapor-liquid mix moves through the pipe lines (15 and 18) and the first drosselling valve (16) into the vessel-separator (19). The first drosselling valve (16) - TEV, controls the intermediate pressure in the vessel-separator (19). From the vessel-separator (19) the cooled liquid moves through the second drosselling valve (21) through the pipe lines (20 and 23) to the entrance of the evaporator (2). The second drosselling valve (21) controls the degree of vapor overheating at the exit (26) of the evaporator (2). From the evaporator (2) the superheated steam moves through the pipe line (10) to the compressor inlet (5).

The air conditioning system is equipped with the device (24), that controls the rotation rate of the engine (6) according to the temperature of the air from the evaporator (2) according to the signal from the temperature sensor (25).

The centrifugal compressor (5) for an air conditioning system (Fig. 2) consists of two centrifugal stages of compression (3 and 4) with sprockets of a pump type (29 and 30) and channel diffusers (31 and 32), two covers - collectors (10 and 11) with nipples of entrance and exit (26 and 14). The gas leaving the diffuser (31) of the first stage of compression (3) gathers in a cover - collector (10) and, then, on channels (13), made in a shield (35) of the first stage (3) and body (9) move in shank bore of the body (9), where the electric motor (6) is installed. Therefrom through the channels (13) of the body (9), the shield of the second stage (36) and insert of the second stage (12) the gases move to the entrance of the second stage (4). Thus, the stages of compression (3 and 4) are bridged among themselves without usage of external pipes. The sprockets (29 and 30) are set on the shaft (37) oppositely. The rotor disks (7) of the electric motor (6) are set on slide fit on the shaft (37) between sprockets (29 and 30). The shaft (37) is established on radial thrust bearings (38), assembled in bearing assemblies. The bearing (38) of the first stage (3) is firmly fixed in the shield of the first stage (35). The bearing (38) of the second stage (4) is a floating support. The inner ring of the bearing (38) of the second stage (4) is firmly fixed on the shaft, and preliminary load is applied to the outer ring, created at the expense of contraction of spiral springs (40) assembled in a sleeve (39). Because of this preliminary

load and the thrust arising from it, the working angle of the radial thrust bearing is set. Besides, the preliminary load compensates for the axial thrust produced by the compressor's operation and directed against the preliminary load. The stator (8) is fixed with adjusting screws in the body (9) and is set in relation to the shield by means of an additional adjusting ring. The channels for passage of the refrigerating gas are made in the rotor and stator disks (7 and 8), and also in the stator frame (8). The design of the compressor is accomplished in such a manner that the gas sucked through the body (9) removes the heat not only from the engine, but also from the bearing assemblies. With the purpose to reduce the axle-load produced by the activity of the compressor, the installation diagram of seals (41 and 42) is selected. The radial grooves of the seal (42) of the second stage (4) are made with a larger diameter, than radial grooves of the seal (41) of the first stages (41). Thus redistribution is reached of the axial and radial component of the moment of the forces which act on the shaft (37) being produced by the sprocket (42) towards the increase of the axial component, which compensates for the axle-load produced by the first stage (3).

In the Fig. 3, the profiles of the sprocket of a pump type and diffuser of a channel type are shown schematically. In general, the idea of creation of the centrifugal compressor for small and middle range of cooling output became real due to creation of a new technique of contouring of the blading section.

The air conditioning system with the centrifugal compressor works as follows (according to the Fig. 1 and 2).

The air conditioning system with the centrifugal compressor works as follows: The unit automatically generates a signal going on the electric motor (6) of the centrifugal compressor (5), the electric power is made, and a compressor's (5) rotor (7) with sprockets (29 and 30) start to be spun. The vapor of a coolant from the evaporator (2) go through the pipe-line (24) on an entrance of a first stages wheel (29), where they are pressed, and warmed. Vapor of a coolant from a sprocket (29) fall in the first stages diffuser (31), in which the kinetic energy will be converted in potential, thus pressure is increased. From the diffuser of the first stage (31) through a special directing detail (34) through channels (13), which are made a shield (35) and body (9), gases move in shank bore of a body (9) compressors (5). Passing through a compressor casing, the cooling agent draws off the heat from bearing assemblies (38) rotor (7) and stator (8). Further, through channels (13), made in a body (9) and shield of a second stage (36), through a special detail (12), the gas moves

on an entrance of a second stage (4), consisting from a sprocket (30), diffuser (32) and special covers –manifolds (11). The processes happening in a second stage (4) are similar to that happen in the first stage (3).

The coolant compressed in a second stage (4) compressors (5), through a nipple (14) is pumped in the condenser (1), in which the heat is drawn off from the coolant at constant pressure. In the condenser happen condensation and sub-cooling of refrigerant. Thus, on escaping of the condenser the overcooled vapor-liquid mix will be derivated and through pipe lines (15 and 18) through a drosseling valve (16) goes in the separator (19). The drosseling valve is controlled by a signal coming from the sensor (17), placed on the exit of the vessel-separator (19). The setup of the drosseling valve (16) is made so that it maintains the necessary intermediate pressure in the cycle. In the vessel-separator there is a process of the vapor-liquid mixture into liquid and gaseous phases. The coolant fluid by the pipelines (20 and 23) through the second drosseling valve (21) moves into the evaporator (2). In the evaporator there is a heat interchange between the coolant fluid and cooled environment (air, water, or other heat transfer medium). There transpires boiling and overheating of the cooling agent. The vapor of the cooling agent is pumped out from the evaporator by the compressor (5). Thus, the cycle closes.

The power of the refrigerating system is smoothly regulated by the change of the consumption of the cooling agent through the system and variation of the rotation speed of the rotor (37) of the compressor (5). The change of the consumption is controlled by the drosseling valve, while the temperature of the cooling agent on escaping of the evaporator serves as the pilot signal for the valve. The signal comes from the temperature sensor (22). The alteration of the speed of rotation of the rotor is controlled by the device (25), while the temperature of the air on leaving the evaporator serves as the pilot signal for it (2), this temperature is defined by the temperature sensor (26).

The pressurized integral centrifugal compressor with the built-in disk electric motor of the present invention, due to the high efficiency of stages of compression, electric motor, effective system of cooling, simplicity of the design and utilization of high speed rolling contact bearings, can be applied, *inter alia* in the following fields:

1) air conditioning systems with the power of 7-10 kW;

2) low power chillers;

3) with a particular change of the blading sections design the compressor can be used as an air compressor, including for turbo-supercharging.

Figs. 4, 5, 6, and 7 represent an illustration of the motor. The openings, or spaces through which the cooling gas may flow are marked therein by arrows.

CLAIMS

1) A refrigerating system comprising an evaporator, a condenser, two diffusers, and a two-stage centrifugal compressor assembly with a built-in valve disk collectorless motor between the compressor stages.

2) A refrigerating system according to claim 1, wherein the flowing gas flowing from the first stage of the compressor to the second stage of the compressor, flows through spaces in said motor, and thus effectively cools said motor.

3) A refrigerating system according to claim 1, wherein said system is equipped with a drosseling valve, which controls the temperature and the pressure of the cooling agent upon leaving the evaporator.

4) A refrigerating system according to claim 1, which is automated in such a way that provides a smooth start and stop of the compressor.

5) A refrigerating system according to claim 1, which is equipped with means of changing the engine speed which controls the temperature of the air on the exit of the evaporator.

6) A refrigerating system according to claim 1, wherein said diffusers are designed to improve their efficiency.

7) A novel sprocket and diffuser substantially as illustrated in figure 3.

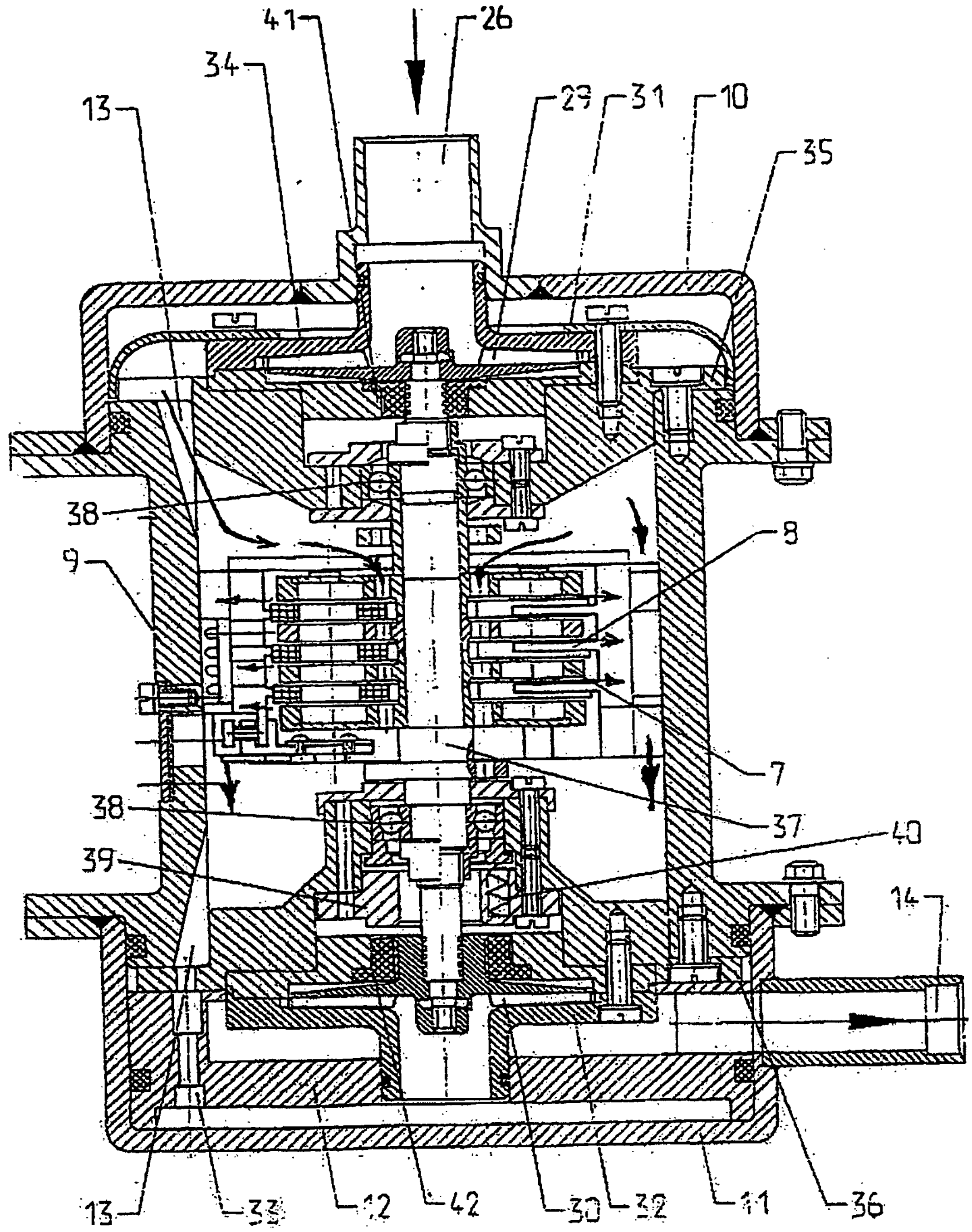


Figure 1

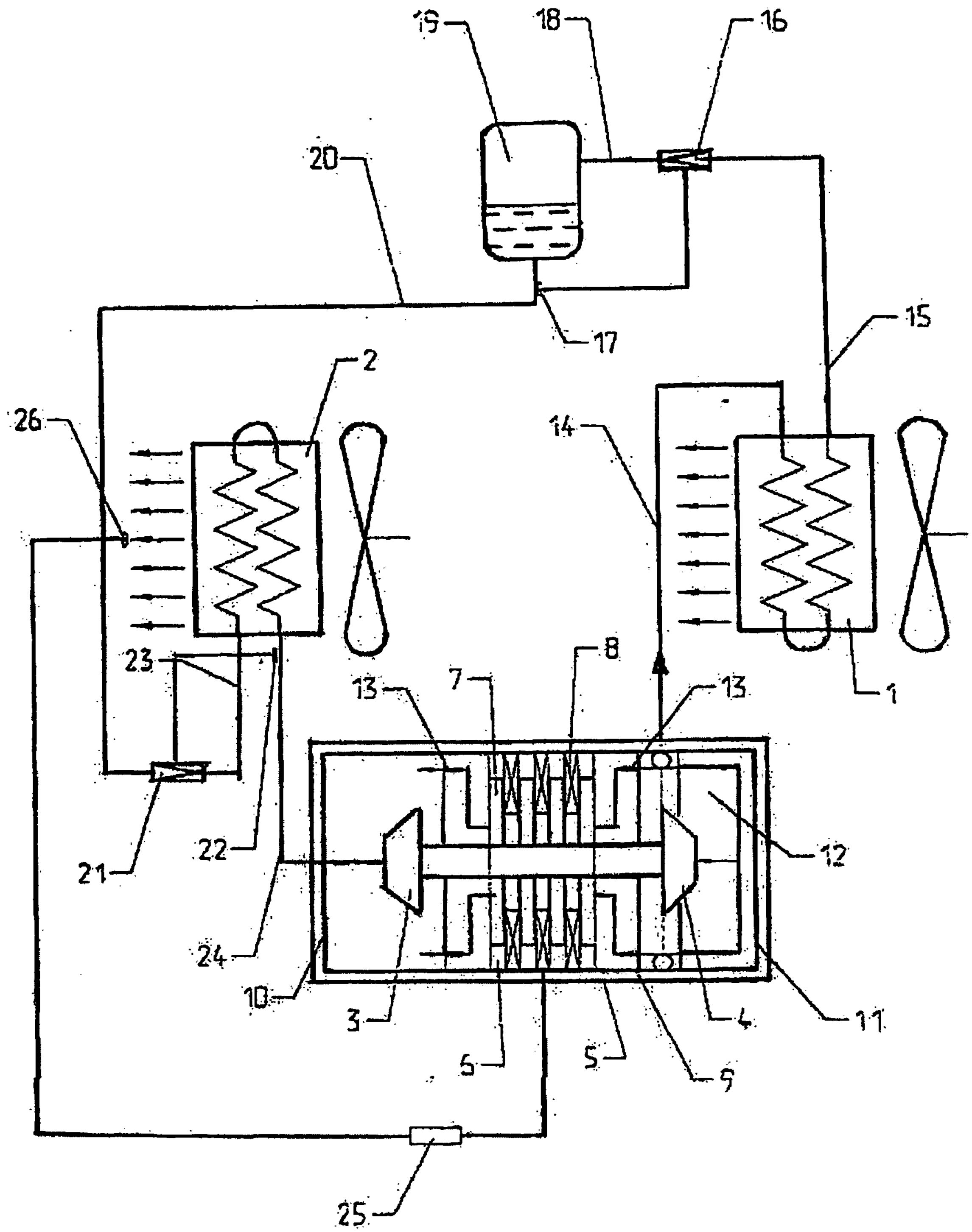


Figure 2

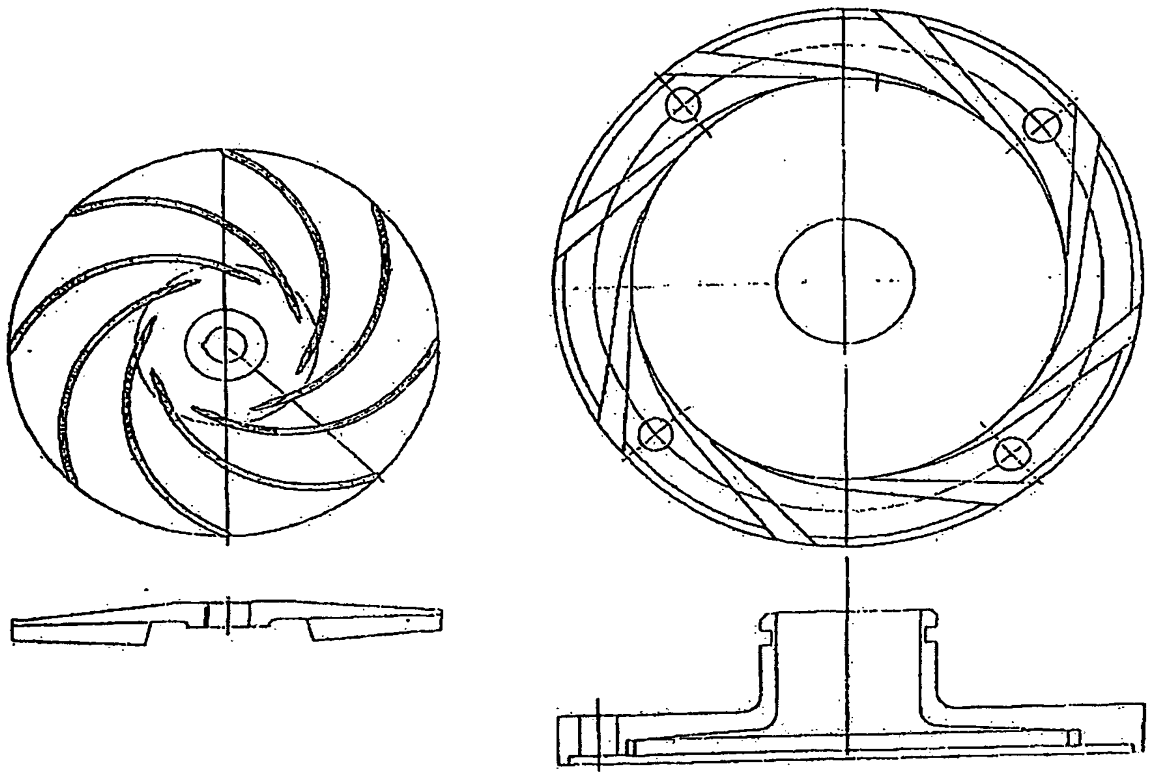
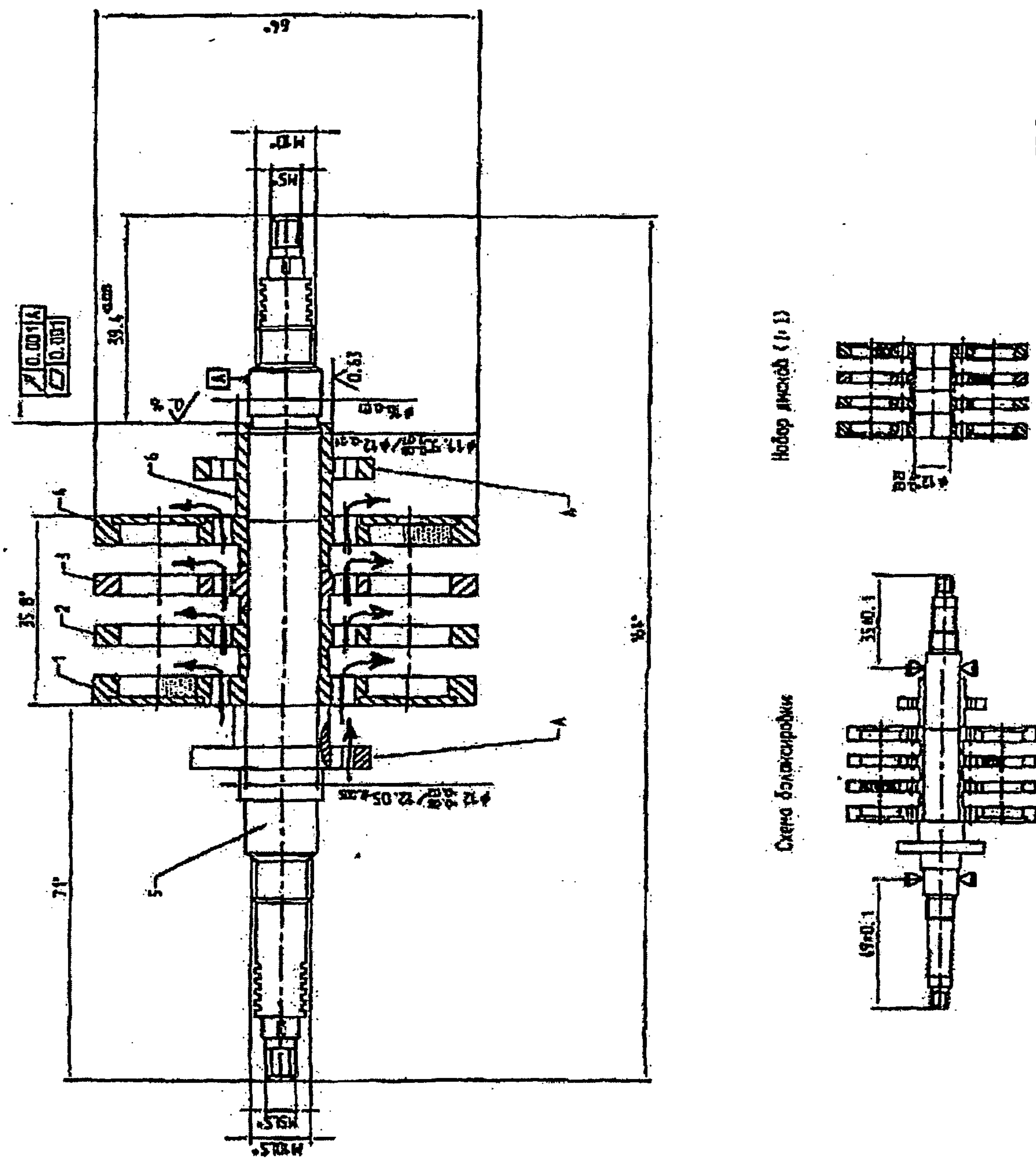


figure 3



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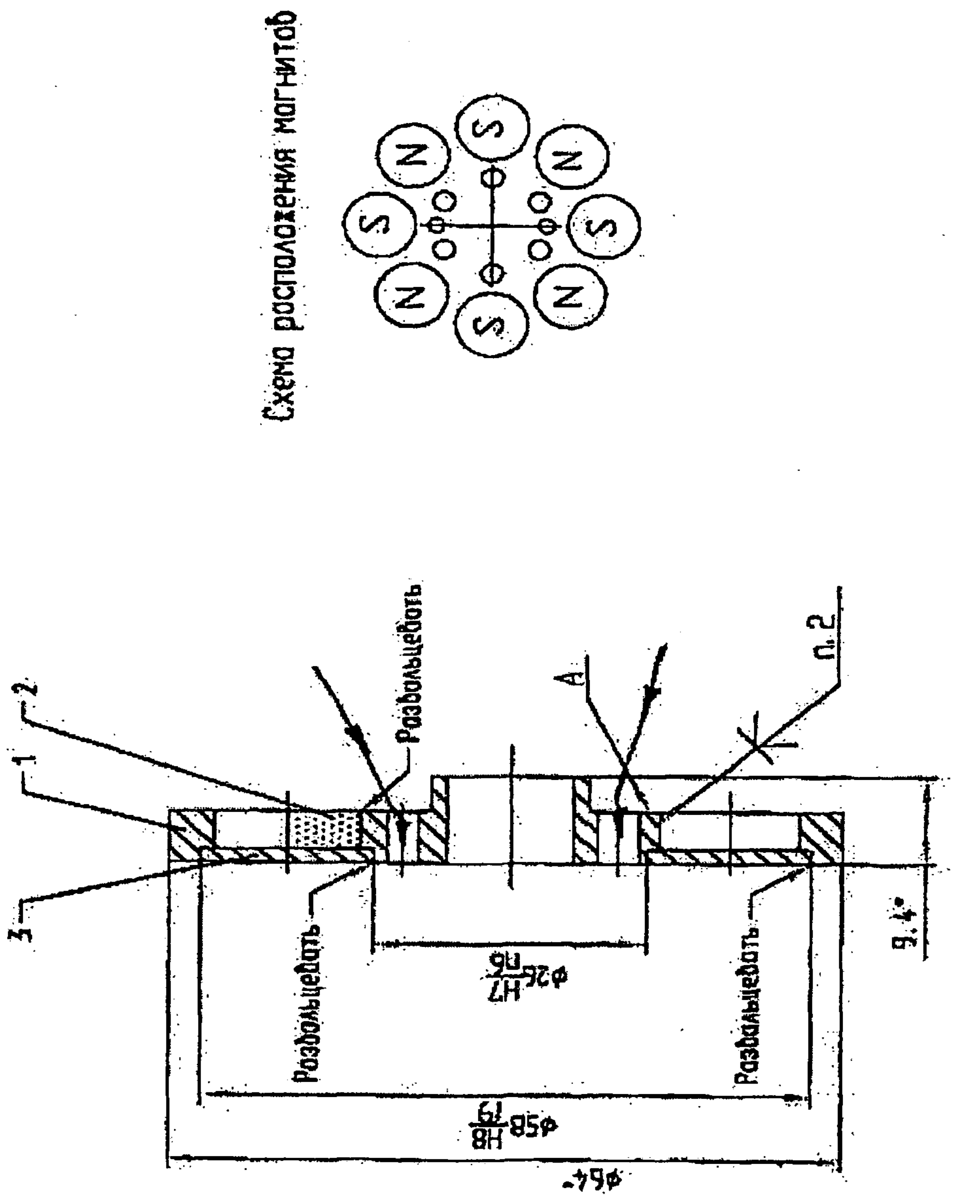
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Figure 5

1. Допускается замена материала на сплав титановый ВТ16 ГОСТ 19807-91 и ВТ14 ОСТ1-90107-73.
2. Отверстия технологические для сборки дисков на валу.
3. * Размеры обеспеч. инстр.

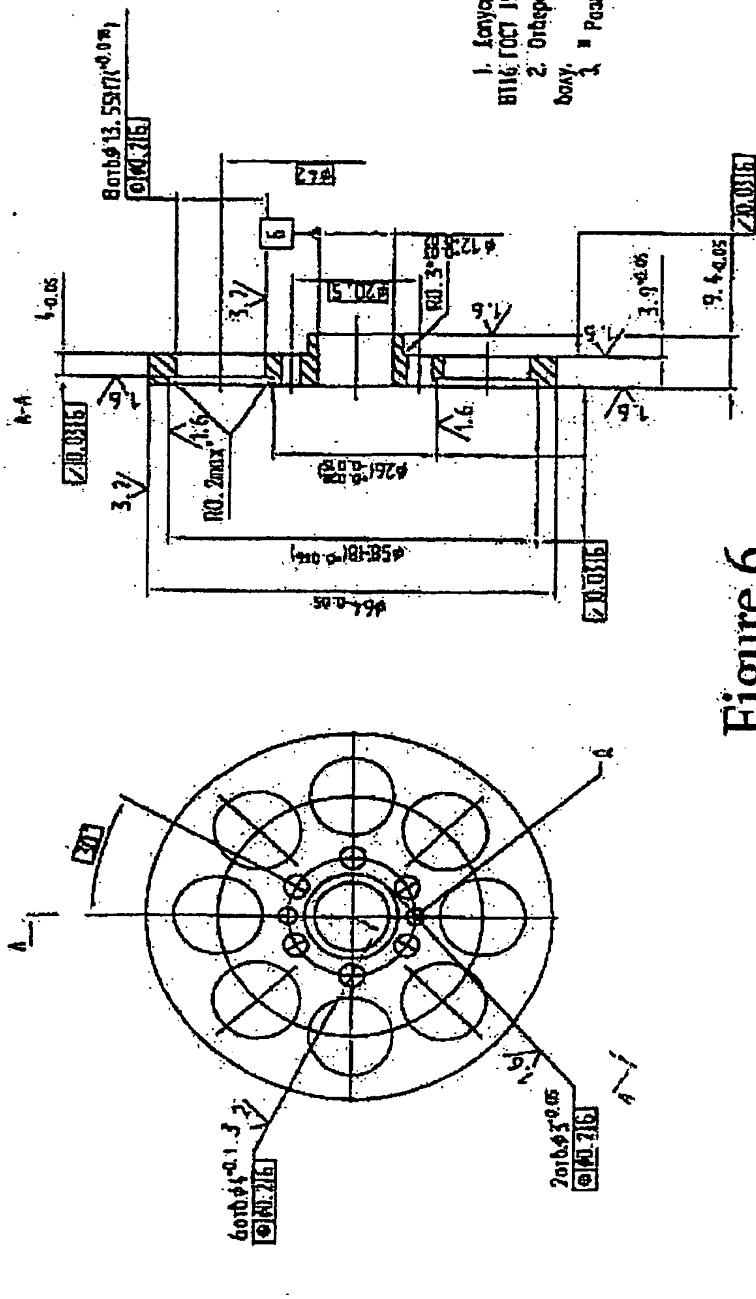

Figure 6

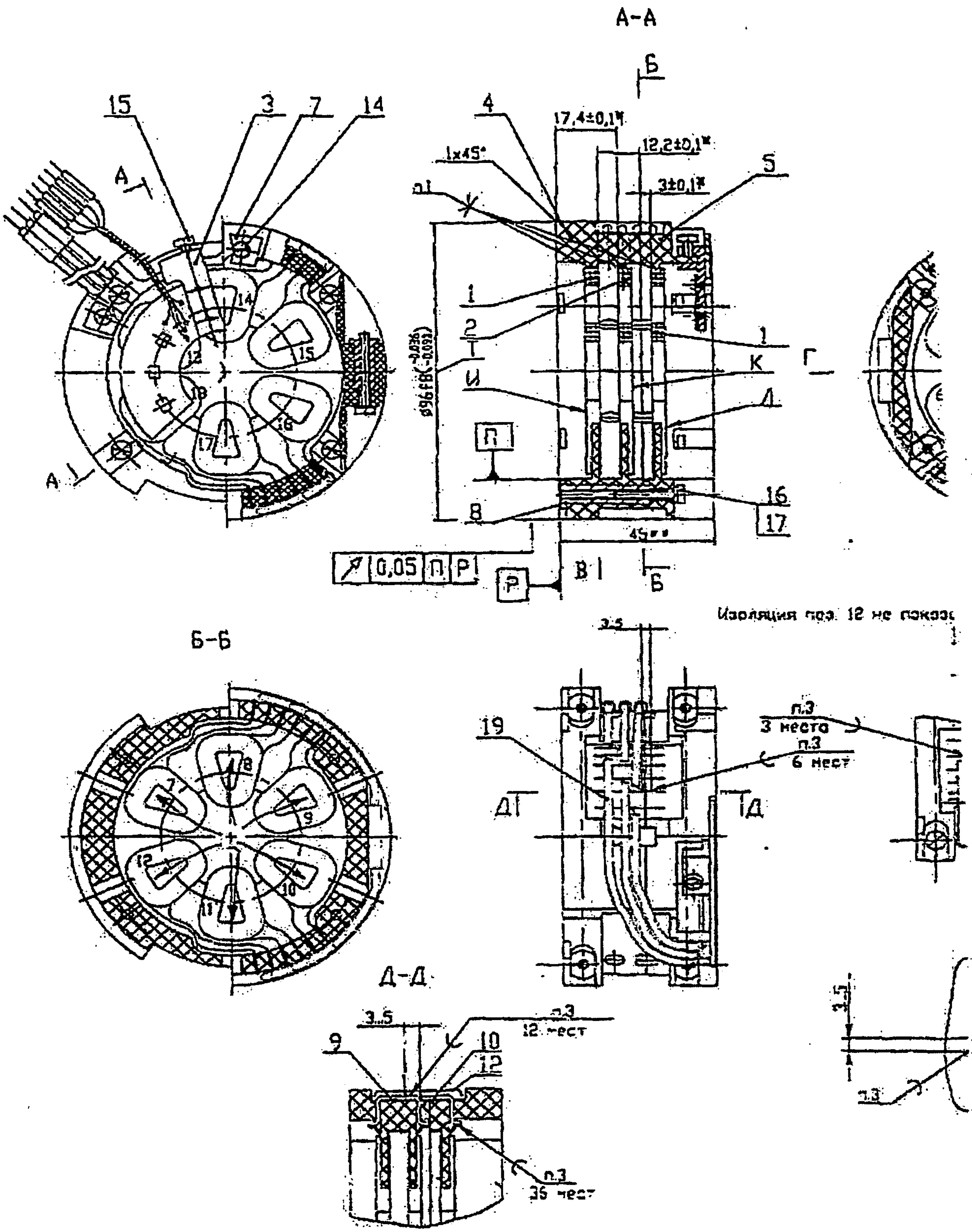
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Figure 7

INTERNATIONAL SEARCH REPORT

national Application No
P[illegible]U 01/00561

A. CLASSIFICATION OF SUBJECT MA[illegible]
IPC 7 F25B1/10 F04D25/06 H02K9/12

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
IPC 7 F25B H02K F04D

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

EPO-Internal

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	US 3 447 335 A (RUFF JOHN D ET AL) 3 June 1969 (1969-06-03) column 2, line 71 -column 9, line 75; figures 1,2,9	1,2
Y	WO 99 65133 A (SHENKAL YUVAL ;SMITH STEPHEN H (US); SMITH TECHNOLOGY DEV LLC (US)) 16 December 1999 (1999-12-16) page 24, line 4 -page 37, line 22; figures 1-3,9,10,12	1,2
A	WO 00 49296 A (KULIKOV NIKOLAI IVANOVICH ;VERESCHAGIN MIKHAIL PETROVICH (RU); KAR) 24 August 2000 (2000-08-24) page 5, line 11 -page 8, line 9; figures 1,2	1,2

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[X] Further documents are listed in the continuation of box C.

[X] Patent family members are listed in annex.

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10 May 2002	23/05/2002

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